PATENT SPECIFICATION

DRAWINGS ATTACHED

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COMPLETE SPECIFICATION

Hydraulic Power Transmissions for Vehicles

We, THE AUSTIN MOTOR COMPANY LIMITED, of Longbridge, Birmingham, a British Company, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to hydraulic power transmissions for vehicles, of the type in which substantially the whole of the power is transmitted hydrostatically by transference of hydraulic fluid between primary and secondary units, and in which a neutral or idling condition is obtained by providing a by-pass, controlled by a valve, across the high and low-pressure sides of the hydraulic circuit, and engagement of drive is effected by moving the by-pass valve gradually to restrict and shut off the by-pass.

In the use of hydrostatic transmissions in road vehicles, it is important to secure smooth starting without excessive pressures or sudden changes in pressure occurring in the hydraulic system, and this is particularly important in the simplest forms of these transmissions; namely, in which substantially the whole of the power transmitted, throughout the starting operation, is transmitted hydrostatically by transfer of liquid between primary and secondary units. It has previously been proposed, in such transmissions, to obtain smooth starting by the use of a by-pass valve connecting the high and low-pressure sides of the hydraulic circuit; this valve being operated by a pressure on one end of it, controlled by a centrifugal valve, driven at a speed proportional to engine speed, so that the opening pressure of the by-pass valve increases quadratically with engine speed.

The purpose of the present invention is to provide an alternatiev starting system for the above-mentioned simplest forms of vehicular hydraulic power transmissions. It is a disadvantage of the arrangement previously proposed that, owing to the quadratic characteristic, the increase in by-pass valve opening pressure is too rapid to give sufficiently smooth and gradual starting in the case of those transmissions. We have discovered, however, that by employing a by-pass valve controlled by a centrifugal-type starting valve arranged to give a much more nearly linear increase of pressure with speed, the above disadvantage may be avoided; and we have devised satisfactory forms of such a centrifugal-type starting valve which (without having to raise the delivery pressure of the associated auxiliary pump) afford linear or nearly linear speed-pressure characteristics that are substantially independent of temperature variations or changes in the viscosity of the working fluid.

According to this invention, in an hydraulic power transmisison for vehicles, of the type specified above, the opening pressure of the by-pass valve is controlled by a centrifugal-type starting valve driven at a speed proportional to engine speed, and this starting valve comprises one or more rotating governor weights acting upon a spring of variable rate, such that its deflection under the centrifugal force of the governor weights or weights is proportional or nearly proportional to the rotational speed thereof and is effective to load and deflect a second spring which loads a pressure-control element of the valve; the fluid-pressure controlled by that pressure-control element actuating the by-pass valve; whereby, over a range of engine speeds appropriate to transmission take-up, the working torque is maintained proportional or nearly proportional to engine speed.

As already indicated, the arrangement is

[*Price 4s. 6d.*]

such that the improved centrifugal valve affords a linear or nearly linear rise in the control pressure and by-pass valve opening pressure over the starting-speed range and so ensures a more gradual movement of the by-pass valve (and consequently a smoother engagement of the vehicle drive) than is attainable with a conventional centrifugal valve having a quadratic characteristic.

The pressure-control element of the improved centrifugal valve is preferably of the ball or piston type, and the variable-rate spring is preferably much heavier than the second spring which applies the resulting force to the control element. The second spring may be of constant or variable rate to suit the specific requirements. In the latter connection, it will be appreciated that if the second spring has a rate which is small compared with that of the first, loads on the second spring will have little effect on the deflection of the first spring, and that, if the second spring is of constant rate, then the deflection of the first spring, which is proportional or nearly proportional to the rotational speed of the governor weight or weights (and hence to engine) speed, will result in deflection of the second spring by an amount which is also proportional to that speed, and thus cause the pressure-control element of the centrifugal valve to be subject to loading which is proportional or nearly proportional to engine speed; so that the controlled fluid pressure of the supply will also be proportional or nearly proportional to engine speed.

Variation of the rates of both springs allows considerable scope in selection of the pressure-speed characteristic of the centrifugal-type valve. Whilst a linear characteristic can be produced, as described, it should also be possible to obtain a characteristic such that the rate of increase of the controlled pressure decreases gradually with increasing speed over the range for which the valve is designed.

Also it will be appreciated that it is only convenient to design for a particular pressure characteristic over a limited range of speeds. For example, where a linear characteristic is required over a speed range of 2:1 (say 500 to 1000 r.p.m.), the change in force due to the governor weights over this range will be little over 4:1 (4:1 or exactly if the weights did not have radial outward movement). Therefore, at 1000 r.p.m. the spring rate requires to be more than four times greater than at 500 r.p.m That is to say, there must be more than four times as many active coils in the variable-rate spring at 500 r.p.m., than there are at 1000 r.p.m. As it is difficult to make variable-rate springs in which the number of active coils varies by a ratio of more than 4:1, or 5:1 over the compression range, a linear characteristic for the valve over a speed range of more than 2:1 would be difficult to obtain.

Three embodiments of the invention are illustrated in axial section in the drawings which accompanied the Provisional Specification.

Referring to those drawings, the valve shown in Fig. 1 comprises a cylindrical valve body 10 which is mounted for rotation about its axis by means (not shown) and is formed with an axial blind-ended bore of two diameters, the larger-diameter portion of this bore being remote from the blind end and constituting a housing for a variable-rate coil spring 11 which operates between an annular shoulder 12, afforded at the junction between two internal diameters of the bore, and an abutment disc 13 which is operated upon by the lever arms 14 of two diametrically opposed governor weights 15 pivotally connected, for radial outward movement at 16, to lugs 17 on the open or right-hand end of the rotary valve body 10. The abutment disc 13 is carried by the right-hand end of a plunger 18 which is encircled by the spring 11 and which, at its left-hand end, is formed with a head 19 which has an all round clearance in the right-hand end of the smaller-diameter portion of the bore. The head 19 affords an annular seating shoulder 20 for one end of a second coil spring 21 the other end of which engages a similar seating on a piston-type valve element 22 which is slidable in the smaller-diameter bore. The chamber afforded between the valve element 22 and the blind end of the bore constitutes a transfer chamber, which is connected, by an inlet port 23, to the pressure supply line (not shown) to be controlled, and, by an outlet port 24, to a reservoir (not shown) from which a delivery pump (not shown) draws its fluid for the supply line.

The valve is shown in the wholly unpressurised condition with the valve element 22 completely closing the outlet port 24. When the delivery pump is started up, its delivery pressure is sufficient to overcome the loading of the spring 21 and cause the valve element 22 to open fully the outlet port 24 so that fluid delivered to the transfer chamber passes therethrough to the reservoir. If now the valve body 10 is driven in rotation at progressively increasing speed, the governor weights 15 come into action to overcome the spring 11, at a rate proportional to variation in rotational speed, and to deflect the spring 21 correspondingly so that it moves the valve element or piston 22 to restrict the outlet port 24 progressively and cause the pressure in the supply line to build up correspondingly, full line-pressure being attained when the outlet port 24 is completely closed by the piston 22. End stops 25 and 26 are provided on the piston 22 and the head 19 of the plunger 18 respec-

tively, to a limit the travel of the piston 22 and of the plunger 18 under the action of the counterweights 15, thereby preventing over-loading of the springs 11 and 21.

The embodiment shown in Fig. 2 is similar to that in Fig. 1, except that the valve element 22 is a ball which controls an axial inlet port 23ˣ instead of the radial outlet port 24.

The embodiment shown in Fig. 3 is similar to that in Fig. 1, except for an axial extension 10ˣ of the valve body. This extension 10ˣ constitutes a housing for a third coil spring 27 which operates upon the levers 14 of the counterweights 15 and in opposition to the spring 11. The third spring 27 is provided to give more scope in designing the valve, and enables linear or nearly linear characteristics to be achieved over a wider range of rotational speeds. A ball-type valve, as in Fig. 2 may be used in this embodiment instead of the piston valve 22.

The second spring in Fig. 1 or Fig. 2 may be either of constant or variable rate to suit design requirements, as may also the second or third spring, or both, in the Fig. 3 arrangement.

In our co-pending Application No. 23423/59 (Serial No. 941288) (from which this Application is divided) we claim: " A rotary pressure relief valve of the centrifugally loaded type characterised in that one or more rotating governor weights, which rotates or rotate with the valve body, acts or act upon a spring of variable rate such that its deflection, due to the centrifugal force on said weight or weights, is proportional to the rotational speed of the latter, and in that said rotating governor weight or weights, or a member having an operative connection therewith, acts or act upon a second spring which applies a resulting controlling force to the movable closure element of the valve ". We make no claim to that valve, *per se*, in the present Application.

We also make no claim to any hydraulic transmission gear of the type that is disclosed and claimed in Specification No. 761,534.

What we claim (subject to the foregoing disclaimers) is:—

1. An hydraulic vehicle-transmission of the type specified, in which the opening pressure of the by-pass valve is controlled by a centrifugal-type starting valve, driven at a speed proportional to engine speed, and the starting valve comprises one or more rotating governor weights acting upon a spring of variable rate, such that its deflection under the centrifugal force of the governor weight or weights is proportional or nearly proportional to the rotational speed thereof and is effective to load and deflect a second spring which loads a pressure-control element of the valve; the fluid-pressure controlled by that pressure-control element actuating the by-pass valve; whereby over a range of engine speeds appropriate to transmission take-up, the working torque is maintained proportional or nearly proportional to engine speed.

2. An hydraulic vehicle-transmission according to claim 1, in which the pressure-control element of the centrifugal-type starting valve is of the ball or piston type, and the variable-rate spring is much heavier than the second spring.

3. An hydraulic vehicle-transmission according to claim 1, having a starting valve constructed and arranged to operate substantially as described with reference to Figure 1 of the drawings accompanying the Provisional Specification.

4. An hydraulic vehicle-transmission according to claim 1, having a starting valve constructed and arranged to operate substantially as described with reference to Figure 2 of the drawings accompanying the Provisional Specification.

5. An hydraulic vehicle-transmission according to claim 1, having a starting valve constructed and arranged to operate substantially as described with reference to Figure 3 of the drawings accompanying the Provisional Specification.

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1 SHEET *This drawing is a reproduction of the Original on a reduced scale*

Fig.1.

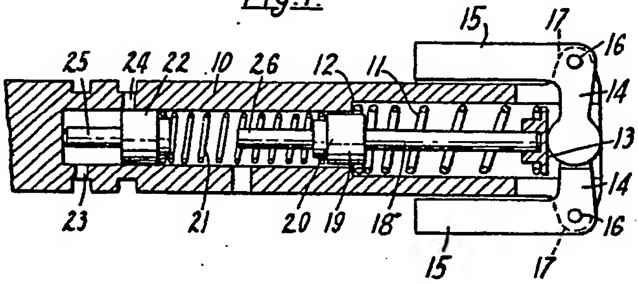

Fig.2.

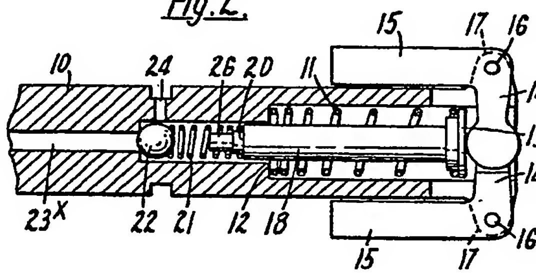

Fig.3.